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A Gas-Steam Combined Cycle Powered By Syngas Derived From Biomass

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Abstract

In this paper, an innovative power plant, constituted by a gas turbine in combined-cycle fuelled by a synthesis gas (or syngas), produced in a local biomass gasifier, is analyzed. The plant is integrated with an external combustion system, fed by cellulosic biomass, connected to a heat exchanger able to increase the air temperature, as in a regenerative cycle. The combustion products pass through a primary heat exchanger placed in the external combustion system, heating the compressed air, which flows into the principal combustion chamber, where a defined quantity of syngas, coming from the gasifier, reacts with the compressed air in a combustion process. The expanded gas, at the turbine exit, before going back into the external combustor, passes through a Heat Recovery Steam Generator (HRSG1) transferring heat to the bottoming Rankine cycle. The superheated steam undergoes an expansion in a steam turbine providing electrical energy. The syngas used in the combustion chamber is produced by a gasification process, based on a Fast Internally Circulating Fluidized-Bed (FICFB). Heat is transferred from the hot syngas (coming from the gasifier) to water, through a second Heat Recovery Steam Generator (HRSG2), producing steam, which is introduced in the gasifier, reacting with the pomace biomass in order to produce the syngas; since the produced quantity of steam is not sufficient for the gasification process, a further quantity of steam is produced in an auxiliary boiler fed by diesel oil, or in different ways, as described in the paper. This kind of plant is especially interesting for regions, like Italian Apulia, where there is a wide culture diffusion for the use of biomass, particularly from olive products, where there are available technologies for use of pruning, virgin and exhausted pomace, and where there are the market conditions for the commercialization of these resources and the incentives available for their energy development. Finally, the overall plant performance is calculated, shown and discussed.

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Keywords: Combined Cycle, Biomass, Syngas, Gasifier

1. Introduction

Unlike other renewables, as wind and photovoltaic, bio-energy sector has a high complexity, due to the need of interaction between the agro-forestry world and industry. The crucial problem is the lack of a biomass market, primarily as a consequence of demand scarcity, and the lack of conversion plants arranged to withdraw a potentially

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Nomenclature

G	$[kg/s]$	mass flow rate
h	$[J/kg]$	specific enthalpy
LHV	$[J/kg]$	lower heating value
k	$[-]$	specific heat ratio
P	$[W]$	power
R	$[J/(kg\,K)]$	gas constant
T	$[K]$	temperature
W_{net}	$[J/kg]$	total specific work
β	$[-]$	compression ratio
ϵ	$[-]$	effectiveness
η	$[-]$	efficiency
c		compressor
fur		external combustion system
gas		gasifier
oil		Diesel oil
reg		regenerator
st		steam
syn		syngas
t		turbine

available energy resource. On the supply side, even if there are considerable quantities of biomass potentially available in many areas, due to high dispersion, absence of rational and efficient collection, packaging, transport and storage systems, limited diffusion of the technological know-how and high costs, their widespread use is not favoured [1]. In this paper an innovative combined cycle power plant (CCPP), fuelled by syngas produced in a local gasifier, fed by olive pomace, is analyzed. An external combustion system, fed by biomass, gives the thermal input to partially heat the compressed air, as in regenerative cycles. This plant is supposed to be located in Apulia, an Italian region, because of the wide culture diffusion for the use of biomass and the available technologies for energy supply using pruning, virgin and exhausted olive oil pomace. The byproducts of the olive oil industry can be distinguished from residual of crops such as olive wood and bush, and residues from the processing of olives. Given the importance of olive cultivation in Apulia, olive pruning residues represent a significant biomass resource. The period in which the waste is actually available goes from January to April and the interventions are applied, from once a year to once every 3-4 years depending on the variety of environmental parameters and level of specialization of the plant. The residue exhausted pomace not reused in the plant is about 70% and, in many cases, is sold as fuel; it also has some marginal application for the manufacture of bricks and woodworks. The peanut, formed from the fraction with a higher content of lignin residue, has a lot of use as fuel in agricultural and domestic boilers, in bakery ovens, etc. This product can also be used to derive the fine cellulose suitable for the manufacture of paints. The use of pruning, due to the fragmentation of the resource in the region and the lack of an adequate level of association and organization of producers, requires efficient solutions to reduce costs during collection, transport and storage.

The purpose of this work is to study an innovative system that fully utilizes the local agro-industrial resources, which in Apulia are: the residue from the production of olive oil, and the waste and pruning of olive trees. To this end, a plant powered almost completely by these types of biomass is considered, evaluating different configurations in order to find the most advantageous one from a technical point of view. The study is complex because of the innumerable possibilities of design choices, hence this work is only a first step in an overall optimization process. The importance of this work is to formulate a simple and flexible mathematical model but able to make a reliable prediction of the performance of the system analyzed in its various configurations. Not having found in the literature studies of similar cases, comparisons will be carried out in terms of performance of different plant solutions, trying to understand the influence of the main parameters.

2. Description of the analyzed plant

An interesting CCPP for the electricity production is analyzed, considering two different plant layouts, namely scheme *a* (Fig. 1) and scheme *b* (Fig. 2). In the plant scheme *a* (Fig. 1), named REXC (Regenerative with EXternal Combustor), an external combustion system, fed by biomass, is able to produce gas at $900 - 1200° C$, which heats the air exiting from the compressor of the gas turbine up to T_5 in a regenerative heat exchanger. Then, in the combustion chamber, a defined quantity of syngas, produced by a gasifier, reacts with the air producing exhaust gas at condition 3. The exhaust gas, expanded up to condition 4 in the gas turbine, flows through a Heat Recovery Steam Generator (HRSG1) heating the operating fluid in the bottoming Rankine cycle of the combined cycle; the cooled gas at the exit of HRSG1, at condition 4′, is then used as comburent air in the external combustor reaching the temperature T_4'', taken equal to T_4. The syngas used in the combustion chamber is produced at high temperature (condition 16 of Fig. 1) by means of a gasifier. The hot syngas is then cooled in a second HRSG (HRSG2), heating a mass flow rate of water from the external condition 13 to condition 15. Finally it is compressed up to the combustor condition. The steam obtained in HRSG2 is then introduced in the gasifier. If this steam is not sufficient for the gasification process, supplementary steam must be produced in a different way. In the present paper three possible solutions to this problem have been considered, indicated as first, second and third solution (see Fig. 3). The first solution is the one used in the representation of both schemes *a* and *b*.

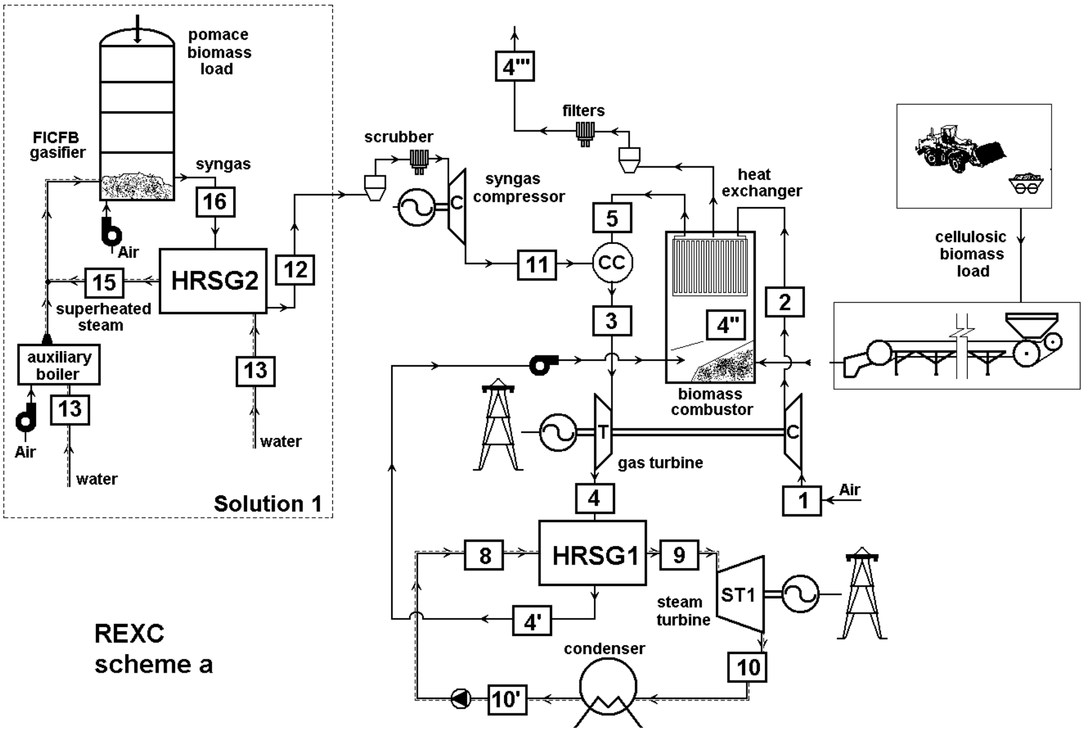

Figure 1: Plant scheme *a* with the first solution for gasifier steam production (REXC)

In the plant scheme *b* (Fig. 2), named CR (Classical Regenerative process), a classical regenerative process is adopted, with a heat exchanger in which the compressed air, at condition 2, is heated up to condition 5 by the exhaust gas coming from the turbine at condition 4. The cooled exhaust gas, at condition 4‴, is sent into the external biomass combustor, where is heated up to condition 4″. In the present paper the temperature T_4'' has been taken equal to T_4. The same three solutions of scheme *a* have been here considered in order to produce the additional steam needed in the gasification process. The overall plant performance depends on the main characteristic parameters of the Joule cycle, namely the compression ratio, β, the maximum cycle temperature, T_3, the machine efficiencies, the air mass flow rate, G_a, and also on the operating conditions of the gasifier and of the external combustor. In the plants here

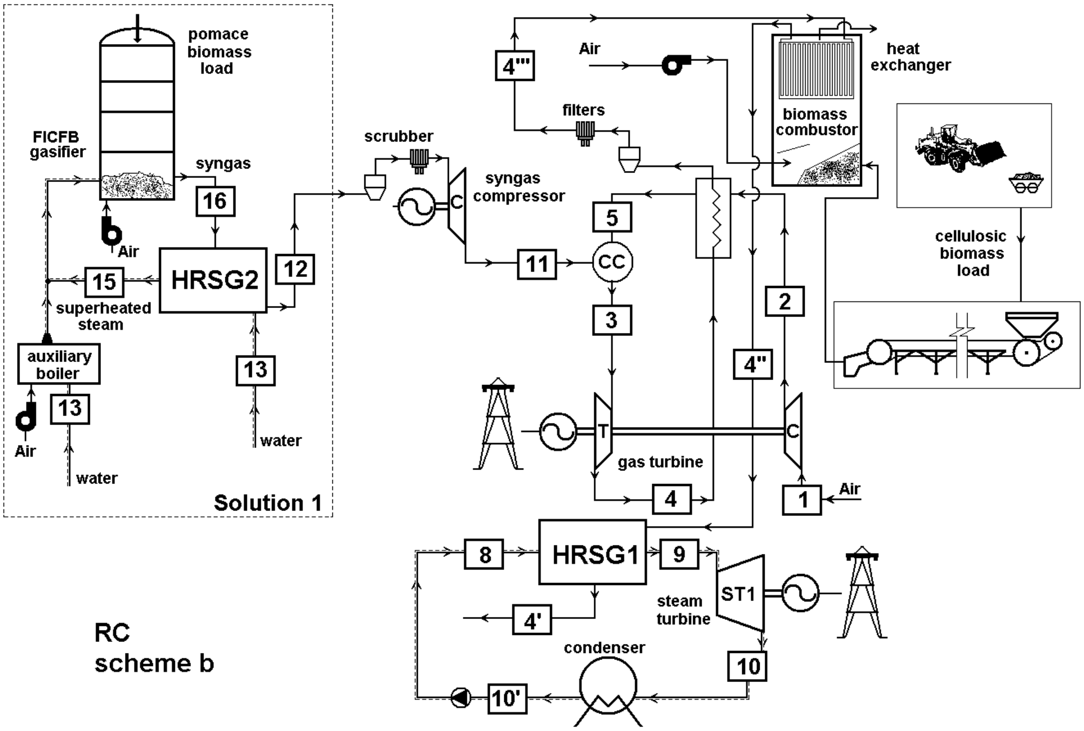

Figure 2: Plant scheme *b* with the first solution for gasifier steam production (RC)

considered, the external combustor is fed by ligno-cellulosic biomass derived by the pruning of olive trees and the gasifier by pomace with less than 15% of humidity. Here, the strategic aspect is the biomass supply, which should be reliable for a long period and at competitive costs.

3. Thermodynamic model of the cycle

Basically, the thermodynamic cycle is the classic Joule-Rankine combined cycle, which is calculated with the assumption of semi-perfect (or thermally perfect) gas, considering the specific heats and the enthalpies variable with temperature in a polynomial way. The analytical models of the various components are now analyzed.

3.1. Gasification plant

The biomass gasifier is based on the Fast Internally Circulating Fluidized-Bed (FICFB) process with sand recirculation, described in several papers (e.g., see [2], [3], [4], [5]). The mass flow rate of syngas, G_{syn}, obtained in the gasification process, depends on the quantity of the fed biomass, G_{pomace}, which consists of exhausted pomace, a residue of the olive oil production. A fixed part of the pomace supply is intended for energy use. In Apulia Region, it is reliable the use, in the gasification process, of a pomace mass flow rate, G_{pomace}, in the range of $(0.05 - 0.25)\, kg/s$. A simple mathematical model of the gasifier has been set up, based on the following equations:

$$\frac{G_w}{G_{pomace}} = 0.90 \tag{1}$$

where G_w is the steam mass flow rate needed by the gasifier, in thermodynamic condition of point 15 of Fig. 1 and Fig. 2. Moreover the gasifier efficiency is given by:

$$\eta_{gas} = \frac{G_{syn}\, LHV_{syn} - G_w\, h_{15}}{G_{pomace}\, LHV_{pomace}} \tag{2}$$

3.2. Heat Recovery Steam Generator for the Syngas (HRSG2)

The Heat Recovery Steam Generator (HRSG2) produces steam by means of heat recovered from hot gases, that, in the case of the concerned installation, is the syngas produced by gasification, before being filtered and compressed. In fact, the syngas has a high enthalpy exiting from the gasifier at a temperature of about 850° C allowing the heat transfer to the water, producing steam. The energy balance of HRSG2, which takes into account the heat exchange between syngas and water, is expressed by means of the following equation:

$$G_{syn}\,(h_{16} - h_{12}) = G_{w2}\,(h_{15} - h_{13}) \tag{3}$$

The steam produced, G_{w2}, is lower than that necessary, G_w, for the operation of the gasifier, and therefore it must be integrated; different solutions may be adopted to integrate the steam, among which the following three have been considered:

1. Solution 1 (Fig. 3.a). The produced steam is integrated, up to the desired flow rate, from an auxiliary boiler, whose combustor is fed by a traditional fuel (G_{oil}) such as diesel fuel, in which water (G_{w1}) entering at ambient conditions (13) is vaporized and superheated up to conditions 15; it is mixed with the steam deriving from HRSG2, having the same enthalpy h_{15}, and then all the steam mass flow rate, G_w, enters into the gasification reactor. The energy balance in the combustor is expressed by the following equation:

$$\eta_b\,G_{oil}\,LHV_{oil} = G_{w1}\,(h_{15} - h_{13}) \tag{4}$$

2. Solution 2 (Fig. 3.b). The steam integration needed for the gasifier, G_{w2}, is obtained by bleeding the steam at condition 9 from HRSG1 and laminating it. In this way, however, the power output of the steam turbine, P_{st1} is lowered.
3. Solution 3 (Fig. 3.c). The superheated steam, G_{w2}, produced by HRSG2 is integrally used in a Rankine cycle, producing an additional mechanical power in a second steam turbine, whereas the steam required for the gasifier, G_w, is totally produced by an auxiliary boiler fed by a traditional fuel. This solution increases the total power of the combined cycle, with the contribution of the second steam turbine (P_{st2}) but also increases the consumption of auxiliary diesel oil, G_{oil}, minimum in the first solution and zero in the second. The plant will be more expensive due to the second Rankine cycle realized with the heat obtained by cooling the syngas.

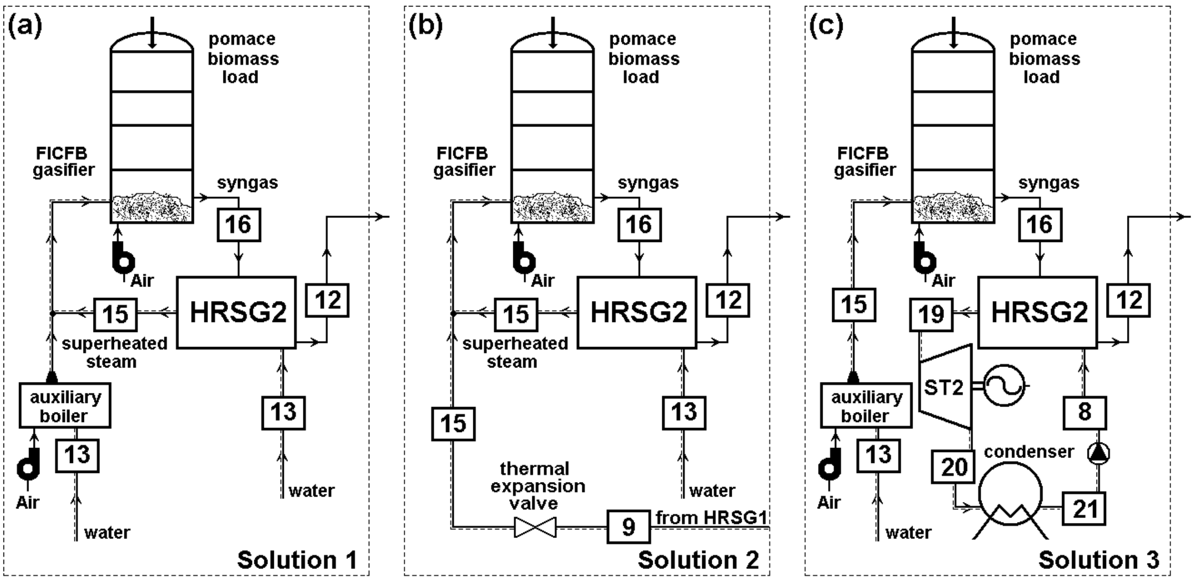

Figure 3: First (a), second (b) and third (c) solution for gasifier steam production both for scheme a and b

3.3. Combustion chamber

Table 1 shows the composition of the syngas produced by steam gasification, taken into account in this paper. Neglecting the small amounts of N_2 and H_2S, its composition can be expressed as follows:

$$\rho\, CH_4 + \omega\, H_2 + \gamma\, CO + \delta\, CO_2 \tag{5}$$

whose Lower Heating Value, LHV_{syn}, is equal to $15200\, kJ/kg$.

Table 1: Composition of syngas

Component	Symbol	Molar fractions (%)
H_2	ω	53.5
CH_4	ρ	7.2
CO	γ	21.1
CO_2	δ	18.1

3.4. External combustion system

The external combustion system in which the biomass is fed into an adiabatic combustor, is able to produce burned gas at high temperature ($900 - 1200°\, C$) (See Ref. [6]). Through a heat exchanger, heat is transferred from the combustor flue gases to air at the compressor exit before entering the main combustor of the plant in scheme *a* (Fig. 1), or to the gas downstream of the regenerator in scheme *b* (Fig. 2).

3.4.1. Scheme a (Fig. 1)

In the REXC (Regeneration with EXtra Combustor) scheme, the external combustor is used to re-heat the gas coming from HRSG1 at h'_4, up to h''_4. The regeneration effectiveness, ϵ_{reg}, is defined as:

$$\epsilon_{reg} = \frac{G_a\,(h_5 - h_2)}{(G_a + G_{syn})\,(h''_4 - h_2)} \tag{6}$$

The efficiency of the external combustion system is given by:

$$\eta_{fur} = \frac{(G_a + G_{syn})\,(h''_4 - h'_4)}{G_{fur}\, LHV_{fur}} \tag{7}$$

Neglecting heat losses to the atmosphere, the heat balance of the regenerative heat exchanger can be expressed as follows:

$$G_a\,(h_5 - h_2) = (G_a + G_{syn})\,(h''_4 - h'''_4) \tag{8}$$

In the present case it has been supposed that $h''_4 = h_4$.

3.4.2. Scheme b (Fig. 2)

In the RC (Classic Regeneration) scheme, the external combustor is positioned downstream of the regenerator, of conventional type, and increases the temperature of the combustion gases from conditions $4'''$ to $4''$. In this latter condition the exhaust gas enter in the steam generator HRSG1. In this case too, h''_4 has been assumed to be equal to h_4. The solving equations are now the following:

$$\eta_{fur} = \frac{(G_a + G_{syn})\,(h''_4 - h'''_4)}{G_{fur}\, LHV_{fur}} \tag{9}$$

$$\epsilon_{reg} = \frac{G_a\,(h_5 - h_2)}{(G_a + G_{syn})\,(h_4 - h_2)} \tag{10}$$

3.5. Heat Recovery Steam Generator (HRSG1)

The HRSG1 is fed by the warm gas at the exit of the turbine at conditions 4 (scheme *a*, Fig. 1) or by the gas heated in the post-combustor in conditions 4′′ (schem *b*, Fig. 2). HRSG1 supplies heat to water producing superheated steam that expands in the steam turbine. The energy balance of HRSG1 gives:

$$(G_a + G_{syn})(h_x - h'_4) = G_{w1}(h_9 - h_8) \tag{11}$$

where $x = 4$ in scheme *a*, and $x = 4''$ in schem *b*. In the present analysis, it has been assumed that $h''_4 = h_4$.

4. Results

The analytical model, that simulates the described systems, includes a system of nonlinear equations, which is solved with a combined method of successive substitutions and some typical methods for the solution of nonlinear systems. Table 2 shows the required data needed by this model for the simulation of the design cycle of the considered system. For the two cases under investigation, the computed main cycle temperatures are shown in Table 3.

Table 2: design data

$P_1 = 1\,bar$	$\eta_b = 0.97$	$T_{16} = 1123\,K$	$P_{15} = 1\,bar$	$\Delta T_{appr} = 70\,K$
$T_1 = 290\,K$	$\eta_{mc} = 0.98$	$T_{15} = 773\,K$	$T_{19} = 850\,K$	$\Delta T_{pp1} = 5\,K$
$\beta = 9$	$\eta_{mc} = 0.98$	$G_{pomace} = 0.15\,g/s$	$P_4 = 1\,bar$	$\Delta T_{pp2} = 30\,K$
$T_3 = 1373\,K$	$\epsilon_{rig} = 0.85$	$P_{10} = 0.05\,bar$	$P'_4 = 1\,bar$	$LHV_{syn} = 15200\,kJ/kg$
$\eta_{yc} = 0.87$	$\eta_{fur} = 0.87$	$P_8 = P_9 = 50\,bar$	$T_{13} = 291\,K$	$LHV_{fur} = 15000\,kJ/kg$
$\eta_{yt} = 0.87$	$\eta_{gas} = 0.85$	$T_8 = 305\,K$		$LHV_{pomace} = 18000\,kJ/kg$
$\eta_b = 0.97$	$P_{13} = 1\,bar$	$T_{13} = 291\,K$		$LHV_{diesel,oil} = 41860\,kJ/kg$

Table 3: Temperatures $[K]$

T_1	T_2	T_3	T_4	T'_4	T''_4	T'''_4	T_5	T_8	T_9	T_11	T_12
290	590	1373	866	397	866	644	835	305	796	577	308

Furthermore, for the different cases considered above, the results in terms of efficiency, η, and total power output, P_{tot}, are shown in tables 4 and 5, respectively.

Table 4: Results: efficiencies of the examined plant schemes

solution	η - scheme *a* (REXC)	η - scheme *b* (CR)
1	0.359	0.470
2	0.360	0.476
3	0.357	0.460

Table 4 shows the values of the overall cycle efficiency for the test-cases considered; the first column refers to scheme *a*, indicated with REXC (Regeneration with EXtra Combustor), whereas the second column refers to scheme *b* indicated as RC (classic Regenerative Cycle); the efficiencies are indicated with $\eta_{1,2,3}$ and defined as follows:

$$\eta_{1,2,3} = \frac{P_{tot\,1,2,3}}{G_{pomace}\,LHV_{pomace} + G_{fur}\,LHV_{fur} + G_{oil}\,LHV_{oil}} \tag{12}$$

where

$$P_{tot} = P_{gt} + P_{st} - P_{c,syn} \tag{13}$$

P_{gt} is the gas turbine net power and

$$P_{st} = P_{st1} + P_{st2} \tag{14}$$

P_{st} is the total power delivered by the steam turbines, being $P_{st2} = 0$ in the first and second case of both schemes. G_{oil} is the flow rate of diesel oil burned in the auxiliary boiler. It can be seen that there is a very small difference between the efficiencies of the 3 sub-cases, the second and third solutions being the most and the least efficient respectively, with a maximum difference of the order of 0.84%. Little more significant are the differences in scheme *b*, where a maximum deviation of 3.4% between the max (Solution 2) and the min (Solution 3) efficiencies can be noted (see Table 4). Remarkable is instead the difference between the efficiencies of the two schemes, namely scheme *a* and *b*, with a variation of approximately 30%.

Table 5: Results: power output of the examined plants

solution	$P_{tot}[MW]$ - scheme *a* (REXC)	$P_{tot}[MW]$ - scheme *b* (CR)
1	1.911	1.911
2	1.860	1.860
3	2.017	2.017

The output powers in the three solution of both schemes, *a* and *b*, are shown in Table 5. The two plants (REXC and CR) provide the same powers, since they are characterized by the same operating conditions in terms of thermodynamic properties upstream and downstream of the gas and steam turbines. A parametric analysis has been performed by varying the gas turbine pressure ratio, β. The difference between maximum (Solution 3) and minimum (Solution 2) powers is approximately 8%. Fig. 4 shows the the overall efficiency for the 3 solutions of scheme *a*; as it can be seen the three curves, $\eta_{1,2,3}$, are very close among them, with maximum difference of 1%. The forth curve refers to scheme *b*, that is to say to a classical regeneration (CR) system, and only to the first solution (Fig. 2), for simplicity. This last value is much higher than the others, with an increment of about 33%. The use of the extra combustor for the regeneration is therefore un-effective; this can be explained giving a look to the discharge temperature of the exhaust gas, which is always higher in scheme *a* with respect to scheme *b*.

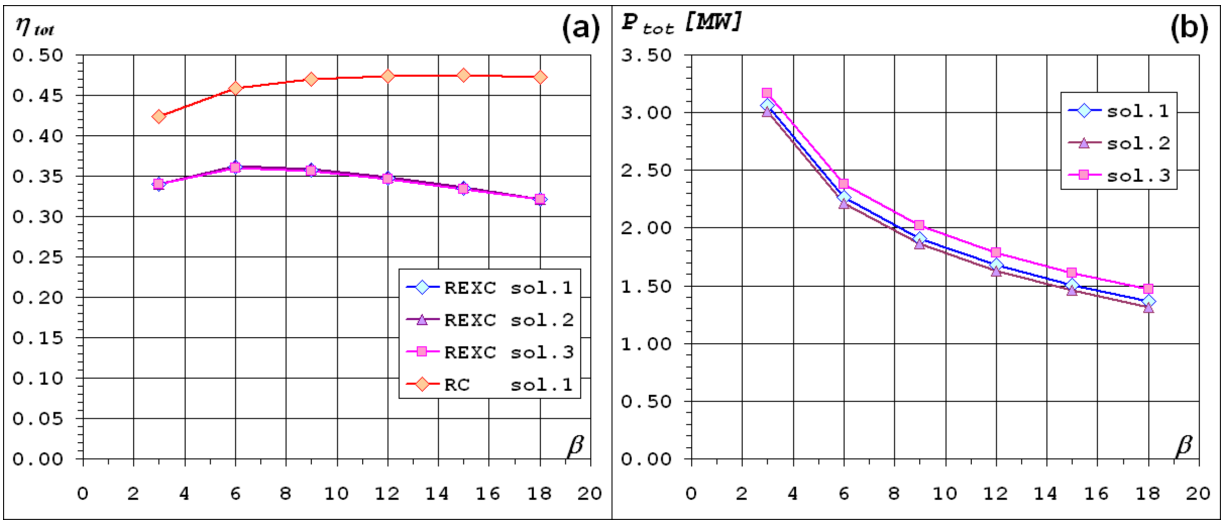

Figure 4: Efficiency (a) and power output (b) of the examined schemes as a function of the pressure ratio

In the REXC case, the curves have a maximum for a β value approximately equal to 8, and a variation of about 11% between the minimum and maximum values. Remarkable, instead is the difference in efficiency compared to the case *b* (CR); the maximum difference between the values of the two curves is approximately 33%. The β for which the CR case has the maximum efficiency is around 12, with a much flatter variation around the maximum. The maximum

β value for this parametric study is equal to 18, since for higher values the conditions compatible with regeneration are not satisfied. Fig. 5.a shows the variation of the total power delivered, $P_{tot\,1,2,3}$, by the cycle in the three conditions. There is no difference between the total power, P_{tot}, delivered by scheme *a* and *b* because:

- the powers output, P_{tot}, of all the schemes and cases are equal depending only on β, T_3, and G_{pomace}, which are the same;
- the powers of the steam cycles are also equal, because the exhaust gas condition upstream (4) and downstream HRSG1 (4) are the same;

From Fig. 4.b it can be seen that the total power output decreases with increasing β, because of the lower temperature of the exhaust gas at the turbine exit, T_4, which determines a lower heat transferred in HRSG1. Among the three solutions 1, 2, 3 the second one delivers the minimum power, whilst the third one the maximum, with a difference of about 8%. Instead, solution 1 gives an intermediate value. In particular, the plant configuration, corresponding to solution 3, provides a higher power output, due to the existence of the second Rankine cycle. However, this scheme involves an economic effort greater than the other two, for both the presence of a second steam turbine and the definitely higher consumption of diesel oil (which is an expensive fuel) with respect to the other cases, since in the auxiliary boiler all the steam, required for the operation of the gasifier, must be generated.

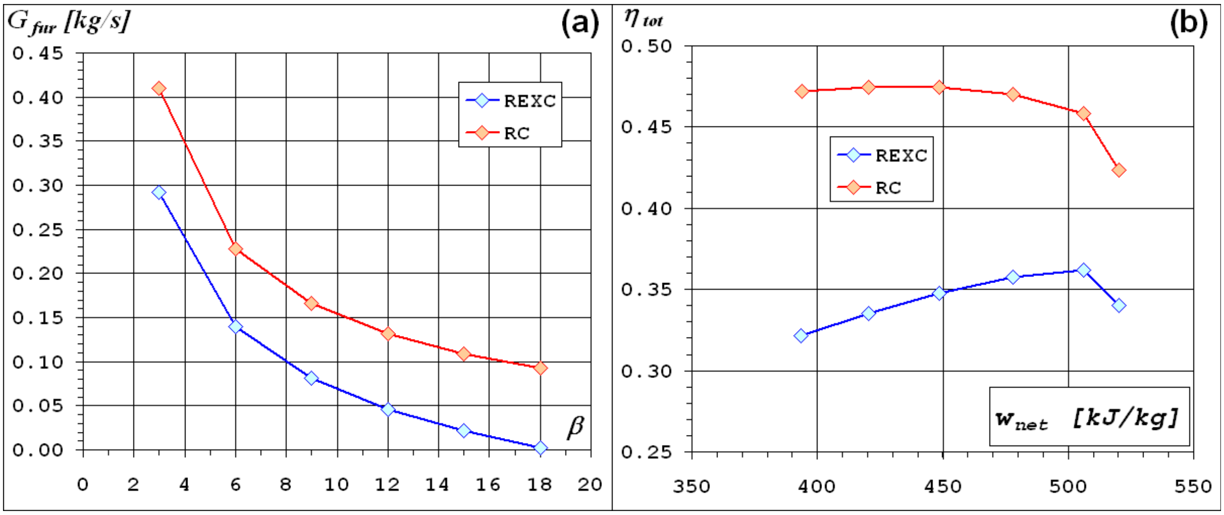

Figure 5: Mass flow rate of biomass fed to the furnace (a) and overall plant efficiency versus specific work at constant T_3 and G_{pomace} (b)

Fig. 5.a shows the furnace mass flow rate, G_{fur}, versus β, with a fixed T_4 value. In scheme *a*, the need of furnace biomass is bigger than in scheme *b*, for the same value of pomace mass flow rate. This difference explains, therefore, the reason why the efficiency of scheme *a* is less than the one of scheme *b*. From Fig. 5.b, which reports the efficiency as a function of the total specific work, W_{net}, it appears that the efficiency of the RC case reaches the maximum value of approximately 0.48, even if a quite low maximum temperature ($T_3 = 1373\,K$), in comparison with the values attainable in conventional plants, and low values of the polytropic turbine and compressor efficiencies, taking into account the small size of these machines, have been considered. These values of the overall efficiency are high and greater than those found in other similar cycles [7].
From this thermodynamic analysis therefore, scheme *b*, known as CR, comes out to be considerably more efficient than the REXC configuration. Furthermore, this preliminary analysis shows that the problem is quite complex and there are many parameters involved in the performance optimization process, both in terms of numerical values and possible configurations. A further study in order to determine an optimum operating condition cannot prescind from economic aspects, not addressed in this work, but that is the natural prosecution of this study.

5. Conclusions

In order to deal with complex plants, such as the ones considered in the present work, the phase of data collection and acquisition of the complex information framework related to bioenergy is of particular importance. The studies carried out have allowed a global energy assessment of a power plant fed with syngas produced in an embedded gasifier and with an external combustor. The gasification process has guaranteed the supply of the syngas necessary for the operation of the combined cycle and it has given an acceptable value of the overall performance, comparable with the performance of this kind of installations. The possibility of using an external moving grate post-combustor was analyzed too. This type of power plant would be able to ensure energy production, necessary for small towns, and meet the requirements for limited geographical areas. It can be assumed that such a system can be located in the province of Bari, in Apulia Region (Italy). The plant under investigation may only operate for 3–4 months a year because the supply of residues are seasonal. It is therefore desirable to provide that the system could also operate with traditional fuels, such as methane, in the rest of the year. Two different plant configurations and three sub-cases each have been analyzed. A simple mathematical model has been developed, that is able to evaluate the performance of the plants treated. From the result analysis, it is evident that between the two proposed plant solutions, called respectively REXC and CR, the latter presents much higher efficiencies, for equal power supplied. The study, which has been carried out in this work, has highlighted the large number of possible hypotheses both in terms of plant layouts and design choices of some important parameters.

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